NASA Technical Memorandum 100800

Test Facilities of the Structural Dynamics Branch of NASA Lewis Research Center

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Summary

The NASA Lewis Research Center Structural Dynamics Branch conducts experimental and analytical research related to the structural dynamics of aerospace propulsion and power systems. The contents of this paper deal with the experimental testing facilities of the branch. Presently there are 10 research rigs and 4 laboratories within the branch. These facilities are described along with current and past research work.

Introduction

The Structural Dynamics Branch at the NASA Lewis Research Center is responsible for conducting analytical and experimental research related to the structural dynamics of aerospace propulsion and power systems. Our research emphasizes the areas of aeroelasticity, vibration control, dynamic systems, and computational structural methods. In the aeroelasticity area, methods for analyzing flutter and force response of bladed-disk assemblies are developed. In vibrational control, methods for controlling vibration of rotors and blades are developed. The dynamic systems area is focussed on systems modeling, microgravity robotics, and traction drives. In computational structural methods, improved algorithms are developed and modern computer principles are exploited to find better ways to solve structural problems.

Our test facilities provide a means of obtaining experimental data which provide the basis for predicting of dynamic phenomena within engines. Each test facility is designed to investigate a specific phenomena. The research conducted using these facilities is applied to support the widest range of end users.

Our research is mostly associated with turbomachinery systems. These experimental facilities are associated with blades, bladed disks, or rotor systems. The experimental research ranges from small bench tests to large-scale wind tunnel tests. The primary emphasis of our experimental work, however, is on relatively small research rigs designed to conduct specific experiments. We currently have 10 research rigs and 4 laboratories:

(1) Transient Blade Loss Rub Rig
(2) Hydrostatic Squeeze Film Damper Rig
(3) Rotating Systems Dynamics Rig
(4) Tapered-Bore Seal Rig
(5) Liquid Nitrogen Eddy Current Damping Rig
(6) Liquid Hydrogen Eddy Current Damping Rig
(7) Dynamic Spin Rig
(8) High Load/Thrust Damper Rig
(9) Hypersonic Engine Seal Rig
(10) Vacuum Roller Contact Rig
(11) Bench Test Laboratory
(12) Magnetic Bearing Laboratory
(13) Robotics Laboratory
(14) Transputer Laboratory

This report describes each research rig and laboratory. For most, the following information is usually given:

(1) Research leader
(2) Rig location
(3) Control room location
(4) Purpose
(5) Principle
(6) Test Apparatus
(7) Additional Information

These experimental facilities are not for our use exclusively. Industrial and academic researchers also use the unique capabilities of these facilities to conduct cooperative research programs of mutual interest.

As mentioned previously, we also conduct experiments in major facilities such as the 8- by 6-Foot Supersonic Wind Tunnel at NASA Lewis. Since we are only users of such facilities and not the responsible organization, they will not be discussed herein.

Transient Blade Loss Rub Rig

Research leader: Albert F. Kascak
Rig location: Building 5, Test Cell SW-18
Control room: Room SW-16

Purpose

The purpose of the Transient Blade Loss Rub Rig (fig. 1) is to investigate the response of a rotating assembly to a sudden unbalance resulting in a rubbing phenomenon. The rig is designed to produce a condition similar to that found when a blade is lost on a small gas turbine engine. Blade tip seal rubs can cause rotor instability. Previous studies have shown that rubs can drive a rotor into a backward whirl and cause it to destroy itself in just a few rotations. Understanding the rub phenomenon can lead to design changes that would result in more fuel efficient engines and engines that are less susceptible to backward whirl instabilities. Recently the rig was used to test active rotor control theories under steady-state, rotor speed transient, and unbalance transient conditions. Piezo-electric pushers were used in a feedback loop to control rotor motion (fig. 2).

Principle

A sudden transient unbalance is used to excite rotor vibrations. The initial unbalance starts when a solenoid activated plunger impacts and removes a balance weight. Figure 3 shows the point of impact of the plunger and weight. The weight is a specially machined 5/16–24 UNF cap screw which simulates a turbine blade. This unbalance causes the rotor vibrations to grow until the disks begin to rub the outer seal segments. With the proper conditions, an unstable backward whirl occurs after a few rotations. The control computer, a Digital MINC-11, senses the rubbing by analyzing channels of transient data.

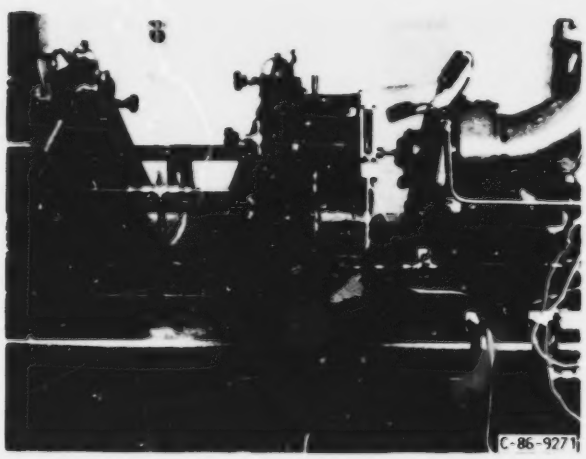

Figure 1.—Transient Blade Loss Rub Rig.

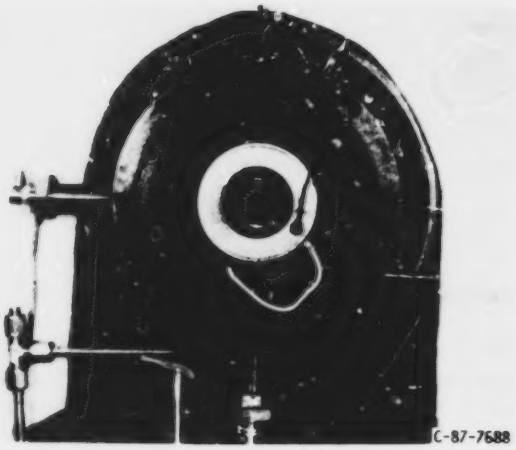

Figure 2.—Piezo-electric pushers used for active rotor control.

Within a few revolutions, a second plunger is activated removing a second weight to rebalance the rotor. Simultaneously, a guillotine is fired severing a 1/8-in.-diameter cable. The cable is used to hold the seal segments in position. The severed cable allows the seal segments to retract from the disks and this prevents the rotor from destroying itself.

Test Apparatus

The rig is composed of an air turbine that is rated at an operating speed of 60 000 rpm. The air turbine is driven by

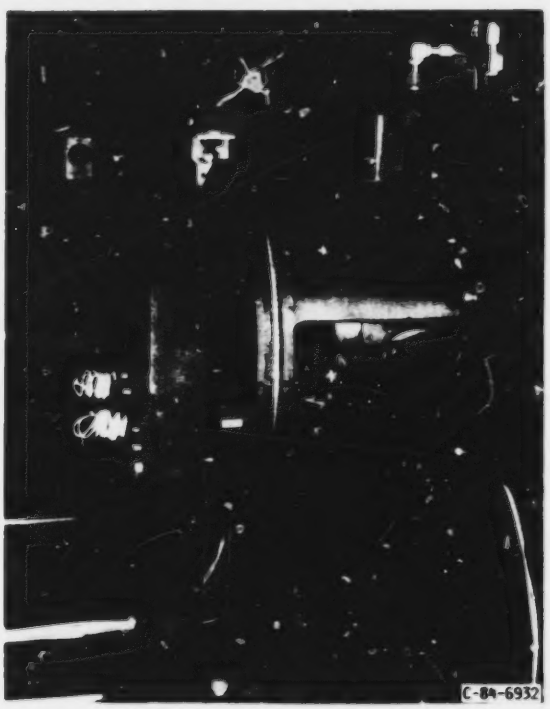

Figure 3.—Blade impact area.

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125 psig shop air. The rig has three rotors that can be interchanged depending on the characteristics of the test. The drive turbine and the two rotor bearing supports are lubricated with an air oil mist system that is pressurized by 125 psig shop air. A squeeze film damper is located at each of the two bearing supports to weaken the rotor motion and the forces transmitted to the bearings. The dampers can be pressurized with air, oil, or an air-oil mixture. There are 10 eddy current proximity probes to measure rotor motions and 4 to measure the damper housing motion. Sets containing 2 probes measure horizontal and vertical motion in a plane. The 14 probes take data at each of 7 planes along the shaft. The orbits of the rotor measured with the probes can be monitored from 6 oscilloscopes in the control room. There are 2 load cells that measure the horizontal and vertical forces at each of the dampers.

The rig has a number of safety features that result in automatic shutdown. Examples of shutdown conditions are an electrical power failure, an air turbine overspeed, and low oil mist pressure supplying the shaft bearings and the drive turbine. The rig will automatically shut down if there is a low oil pressure supply for the squeeze film dampers. The rig is covered with a protective shield, which is a box that is constructed with a wood interior and a steel exterior. The shield is for protection from the loss of the blades and the severing of the wire rope during testing.

Additional information is given in references 1 and 2.

Hydrostatic Squeeze Film Damper Rig

Research leader: Albert F. Kascak
Rig location: Building 5, Test Cell SW-18
Control room: Room SW-16

Purpose

The purpose of the Hydrostatic Squeeze Film Damper Rig is to better understand the vibrations of a transmission incorporating hydrostatic dampers. The system is highly nonlinear and presently there are no reliable prediction methods available. The testing will be used to obtain combined lateral and torsional vibration data to verify codes that are being developed.

Principle

The rig is used to simulate a condition similar to that found in a helicopter transmission. Shaft vibrations are generated from the meshing of the gears and/or a controlled unbalance. The tests involve the running of different gear sets with various gear errors at different torques and unbalance. Tests can be run with or without hydrostatic dampers. The meshing of gears creates vibrations that are transmitted by a shaft, through bearings, into a case. The vibrations are suppressed by hydrostatic dampers that are placed between the bearings and the case.

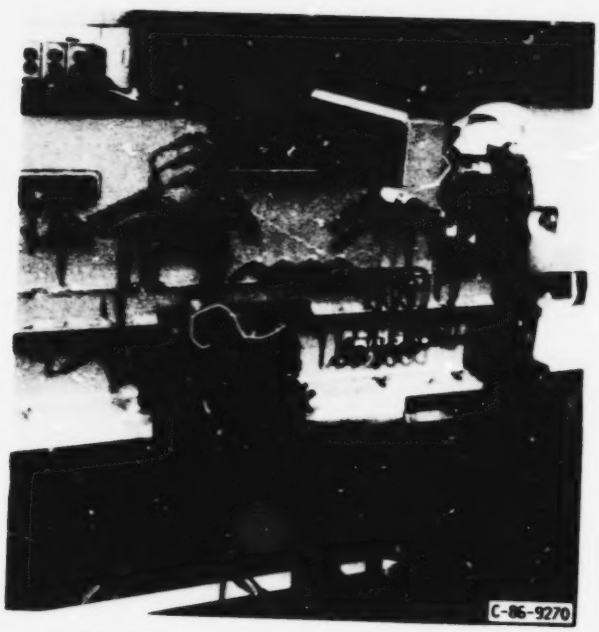

Figure 4.—Hydrostatic Squeeze Film Damper Rig.

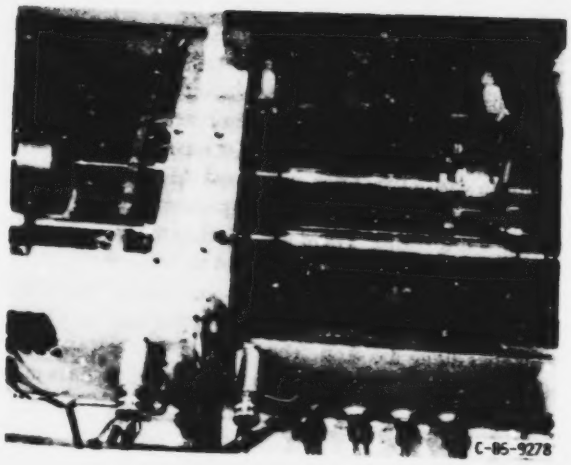

Figure 5.—Shaft system.

Test Apparatus

Figure 4 shows the major components of the test rig which consists of a 10-hp variable-speed dc motor, the transmission, and its casing. A 3000-psi Racine hydraulic package system for squeeze film dampers is located in the basement of the test cell. The bearings are lubricated by a 1-hp ac General Electric motor and a Viking scavage pump. The maximum output speed of the motor is 3500 rpm. The rig contains a low-speed and a high-speed shaft system which is shown in figure 5. The high-speed shaft system is the result of a specially designed spur gear. The lateral shaft vibrations are measured by 16 eddy current proximity probes. The torsional shaft deflections are measured by four BEI Electronics absolute position encoders that are mounted at the ends of each shaft. The maximum operating speed of the rig is established by the maximum operating speed (16 000 rpm) of the encoders. The gear and shaft assemblies are contained in a protective vessel. The hydrostatic damper rig has a number of abort safety systems and alarms such as redundant over-speed trips, low liquid level, thermocouple shutdowns, vibration transducer, and oil flow trips.

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Rotating Systems Dynamics Rig

Research leader: Eliseo DiRusso
Rig location: Building 5, Test Cell SW-14
Control room: Room SW-16

Purpose

The purpose of the Rotating System Dynamics (RSD) Rig is to determine the dynamic characteristics of various rotating systems. The rig permits study of coupled bladed-disk, shaft, and flexible support vibration. The RSD Rig shown in figure 6 is a general facility that can be used in a wide range of research areas such as:

(1) Active rotor control
(2) Steady-state dynamic analysis verification
(3) Transient analysis verification
(4) Coupled blade-disk shaft dynamics
(5) Shaft whirl mechanics
(6) Determination of dynamic bearing stiffness
(7) Force transmission from rotating to static members

Current research with the RSD Rig is focused on demonstrating the potential of active rotor control methods to reduce shaft vibrations when passing through critical speeds and, in general, to improve the dynamic stability of the high-speed lightweight shafting systems that are anticipated for the future. A typical active rotor control configuration is shown in figure 7.

Principle

Active rotor control is a means of limiting rotor vibrations of flexibly supported rotors by use of an electronic feedback control system. The feedback control system senses the vibration of a rotor system and provides damping forces so that the rotor vibration level is maintained within acceptable limits. The rotor vibration is sensed at various locations on the shaft and/or the bearing housing by displacement measurement probes and/or accelerometers. These signals are sent to an electronic controller which processes the signals and commands the force exciters to provide the desired level of damping force to the bearing support. The electronic controller is programmed by entering feedback coefficients to provide any desired amount of damping. These coefficients, which are based on a modal analysis of the rotor system, are calculated by a computer program for various vibrational modes of the rotor.

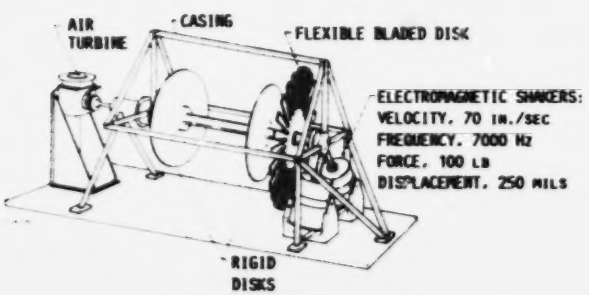


Figure 6.—Rotating Dynamics Rig.

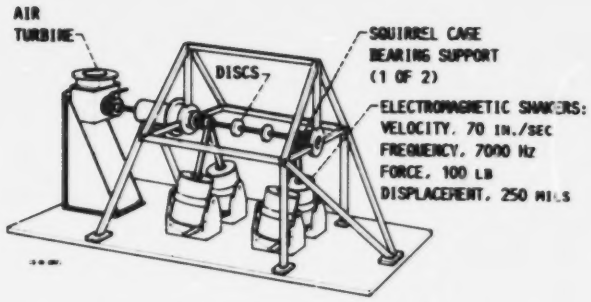


Figure 7.—Active rotor control configuration.

Test Apparatus

The significant features of the RSD Rig are four electrodynamic shakers, electronic controller, rotor shaft, and flexible support frame. An air turbine is used to drive the various shafts. The shakers can simulate various engine operating conditions. The shakers are driven by a signal generator to produce force input to the system under test. Two rotors are available. One rotor (fig. 6) simulates the mass and stiffness of a large turbofan engine rotor and is mounted on a flexible structure which simulates the engine case stiffness. This rotor is limited to 6000 rpm. The other rotor (fig. 7) has three disks and is used for active rotor control experiments. This rotor is supported by squirrel cage springs and is limited to 10 000 rpm.

Additional information is given in reference 2.

Tapered-Bore Seal Rig

Research leader: David P. Fleming
Rig location: Building 5, Test Cell SW-14
Control room: Operated from the test cell

Purpose

The purpose of the Tapered-Bore Seal Rig (fig. 8) is to measure static and dynamic stiffness of a tapered-bore seal with air as the sealed fluid. This rig has also been used to determine the effect of herringbone-grooved shafts with plain bearings on half-frequency whirl.

Principle

Air is supplied to the center of the seal (see fig. 8) and flows out at each end. The pressure at the seal entrance is measured in the ring on the outer diameter of the seal insert. Several shafts are available with varying diameters.

For steady-state data, the shaft is horizontal (fig. 8); a pneumatic loader loads the shaft upward. The air pressures to the seals are set at the desired value. The loader pressure is set so that the applied upward force will just balance the weight of the shaft. The resulting shaft position is recorded and all subsequent shaft motion is referenced to this zero net load position. The shaft speed is then set, and the load is increased in small increments to some maximum value. This load value is just enough to cause contact between the seal and the shaft when the shaft is stationary and to maintain a small clearance when the shaft is rotating. After reaching the maximum value, the load is decreased in increments to the zero-net-load value. The output variable is the seal stiffness defined as the applied force divided by the shaft displacement. The seal leakage is not measured.

The rig was also used to determine dynamic stiffness and damping. In this case, the rig is turned so the shaft is vertical

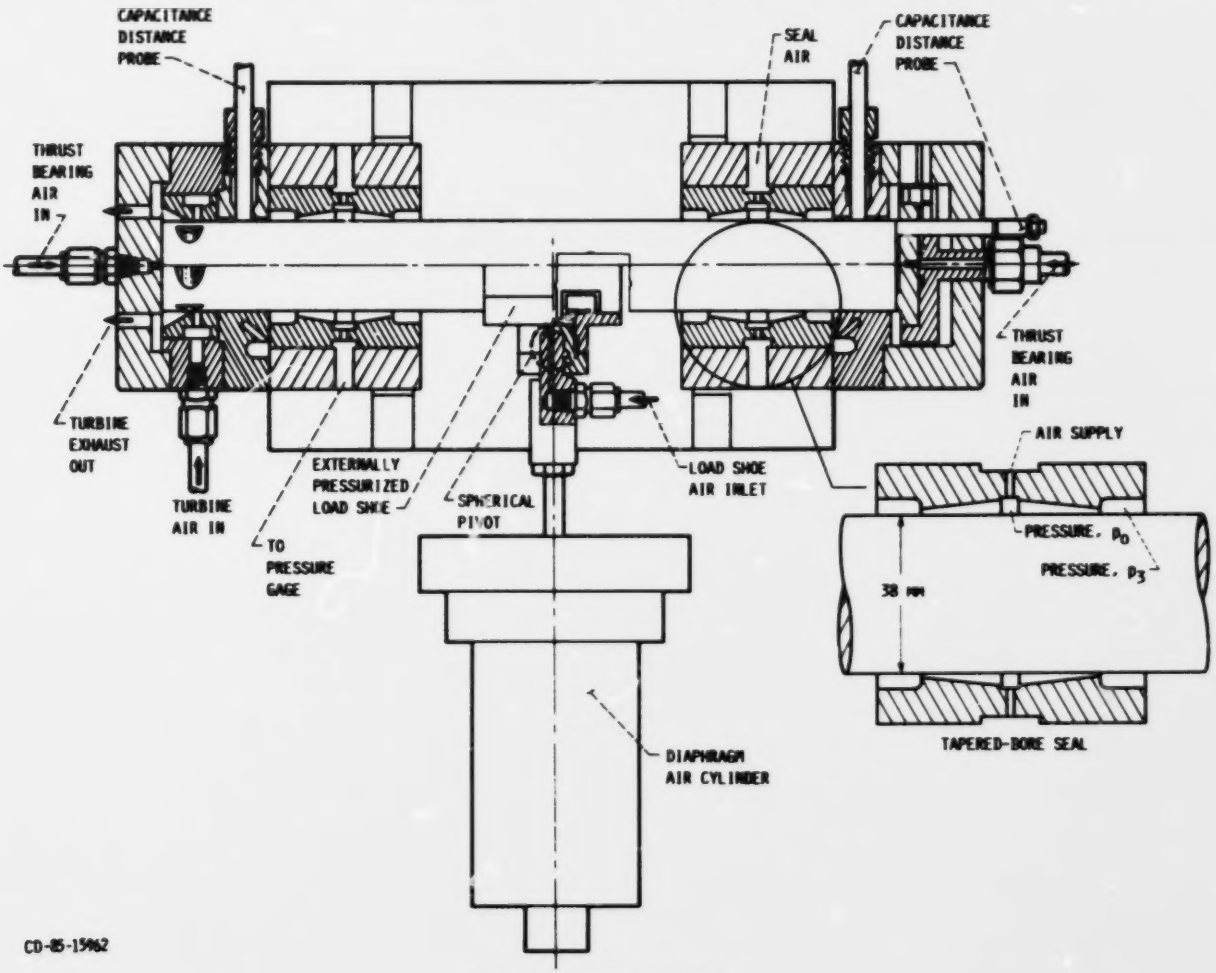
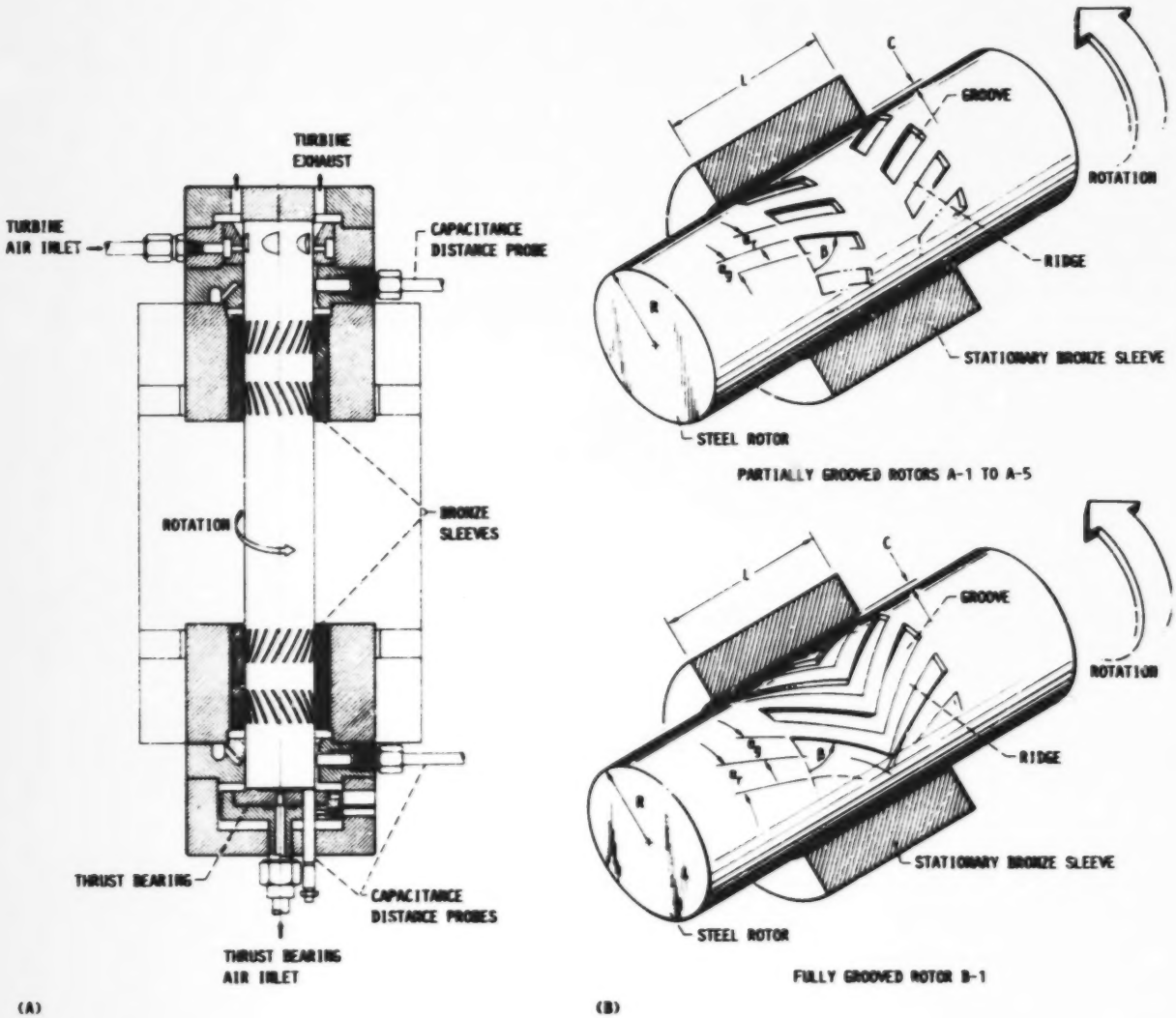


Figure 8.—Tapered-Bore Seal Rig.



(a) Schematic drawing of test apparatus.
(b) Herringbone-grooved bearings.

Figure 9.—Schematic of Tapered-Bore Seal Rig using herringbone-grooved bearings.

and excitation is provided by an unbalance. The capacitance probe signal is routed to a digital vector filter which determines the amplitude of motion and phase angle relative to a reference on the shaft.

Test Apparatus

An air turbine drives a gas bearing or tapered-bore seal rotor assembly at speeds up to 50 000 rpm. For the most recent work, the bronze hydrodynamic bearing sleeves were replaced by the tapered-bore seal. The rotors are mounted and operated in a horizontal position for steady-state data. An upward radial load was applied to the shaft by a rolling diaphragm air cylinder acting through a externally pressurized load shoe. The load cylinder is calibrated to determine the relationship between the cylinder pressure and the applied force. During testing the pressure is measured by a transducer whose output is transmitted to a modular instrument computer (MINC). Two orthogonally oriented capacitance distance probes are mounted outside each seal pair. They are used to measure the shaft under load and to measure the assembled clearance of the shaft in the seals. The capacitance probes outputs were also transmitted to the MINC. The data are read under the control of a Fortran computer program, which provides near-instantaneous reduction of the raw data. The rig has failsafe shutdowns in the event of a electrical power failure or a turbine overspeed. The person in charge of operations is permitted in the test cell during testing provided that the proper ear protection is worn. The

operator must also position himself so that 1/2-in.-thick shield is between him and the running rig.

Additional Information

Tapered-bore seals have been used successfully where straight bore seals have been unsuccessful. One such case is the hot gas seal in the space shuttle's high-pressure oxygen pump. The low centering forces developed by the straight bore seal allowed rubbing to occur as the seal attempted to follow the shaft; this resulted in rapid wear of the seal. The higher film force of the tapered bore seal permitted the seal to follow the shaft motions without rubbing, hence eliminating wear.

In the past, the rig (fig. 9) was used with herringbone-grooved shafts running hydrodynamically in plain bearings. Herringbone-groove geometries and clearances were varied to determined their effect on half-frequency whirl. Herringbone-grooved bearings were considered for space turbomachinery which used a closed-loop Brayton gas cycle. The cycled gas is used as a lubricant making these machines more compact and lighter than machines using oil lubricants. Components like oil pumps, contact seals, oil scavenging, and separating systems that are normally required with conventional lubricated bearings are eliminated.

Additional information can be found in references 3 and 4.

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Liquid Nitrogen Eddy-Current Damper Rig

Research leader: Eliseo DiRusso
Rig location: Building 5, Test Cell SW-1
Control room: Room SW-3

Purpose

The purpose of the Liquid Nitrogen Eddy-Current Damper Rig (fig. 10) is to evaluate eddy-current damping as a means of reducing lateral rotor vibrations at cryogenic temperatures. The rig is used to study and develop rotor damping methods for cryogenic turbopumps such as those used in the space shuttle's main engine.

Principle

The eddy-current damper consists of a copper conductor (99 percent pure copper) and permanent magnets. Figure 11 shows the arrangement of the copper conductor and magnets. Vibration of the copper conductor causes the conductor to cut the magnetic flux lines of the magnet, thereby generating eddy currents in the copper conductor. This process of generating eddy currents dissipates the energy of the vibration and acts as a vibration damper. The damping is proportional to velocity.

The effectiveness of the eddy-current damper is dependent on the temperature of the copper conductor because the electrical resistance of the copper conductor is reduced as the temperature is reduced. This lower resistance results in increased damping. This makes the eddy-current damper especially useful in cryogenic turbopumps because of the low inherent temperatures.

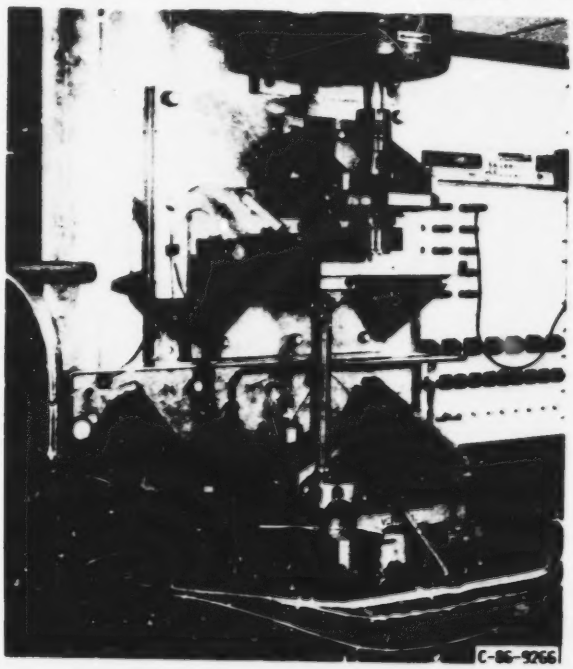

Figure 10.—Liquid Nitrogen Eddy-Current Damper Rig.

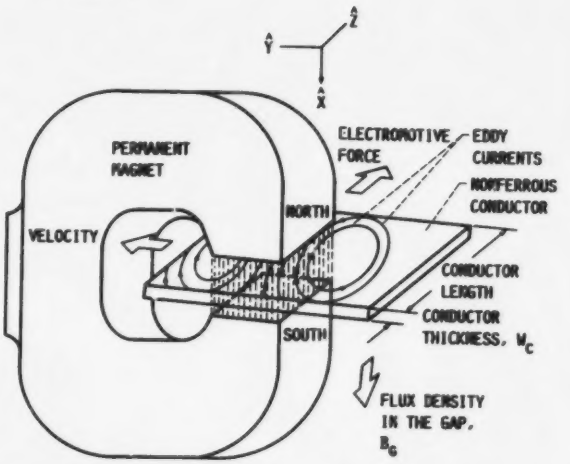

Figure 11.—Schematic of an eddy-current damper showing induced eddy-current paths in the conductor.

Test Apparatus

The test rig (fig. 12) consists of a slender stainless steel shaft which is supported vertically by two angular contact bearings. The lower of these bearings is supported by a squirrel cage spring so that the bearing mount stiffness can be controlled. The copper conductor, which is fastened to the flexible rotor bearing support, vibrates when the rotor passes through a critical speed. The shaft is driven by a 3-hp frequency-controlled ac motor and operates at speeds from 800 to 12 000 rpm. The first shaft bending critical speed is at approximately 6500 rpm. The lower bearing and damper assembly are enclosed in a vessel which contains liquid nitrogen. Thus the bearing and damper assembly are submerged in liquid nitrogen during tests. The liquid nitrogen maintains the damper temperature at −321 °F and simulates temperatures found in cryogenic turbopumps.

A controlled rotor vibration is induced by unbalancing the rotor disk with set screws placed near the outside diameter of the disk. Two proximity probes located near the plane of the lower bearing measure the shaft orbit. Two accelerometers mounted at the lower end of the bearing housing measure bearing housing accelerations in orthogonal directions. Data from the proximity probes and accelerometers and a synchronous (1/revolution) pulse can be recorded on tape. These data can then be processed in a computer which gives synchronous rotor response and phase angle at the lower bearing.

Additional information is given in reference 5.

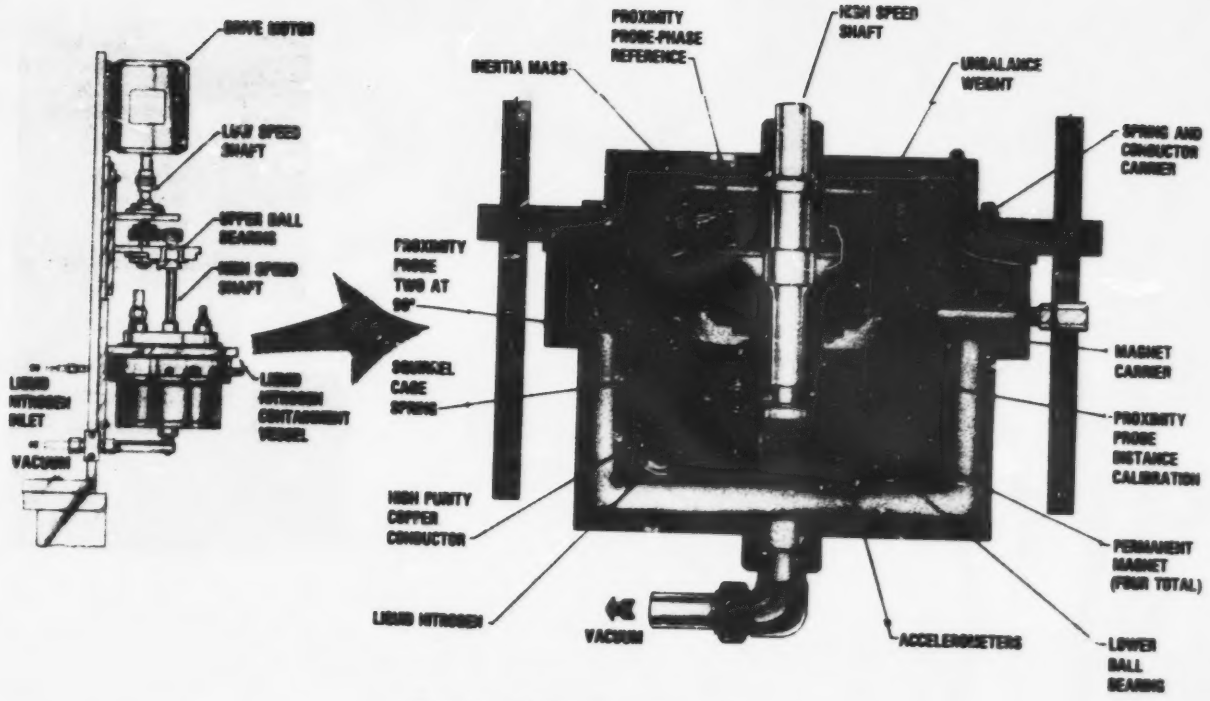

Figure 12.—Eddy-current damper test apparatus.

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Liquid Hydrogen Eddy-Current Damper Rig

Research leader: Eliseo DiRusso
Rig location: Building 203, Test Cell 1
Control room: Same

Purpose

The purpose of the Liquid Hydrogen Eddy-Current Damper Rig (fig. 13) is to evaluate eddy-current damping as a means of reducing rotor vibrations at liquid hydrogen temperatures (−424 °F). The rig is used to study and develop rotor damping methods for cryogenic turbopumps such as those used in the space shuttle's main engine.

The eddy-current damper consist of a copper conductor (99 percent pure copper) and permanent magnets. Figure 11 shows the arrangement of the copper conductor and magnets. Vibration of the copper conductor causes the conductor to cut the magnetic flux lines of the magnet, thereby generating eddy currents in the copper conductor. This process of generating eddy currents dissipates the energy of the vibration and acts as a vibration damper. The damping is proportional to velocity.

The effectiveness of the eddy-current damper is dependent on the temperature of the copper conductor because the electrical resistance of the copper conductor is reduced as the temperature is reduced. This lower resistance results in increased damping. This makes the eddy-current damper especially useful in cryogenic turbopumps because of the low inherent temperatures.

Principle

The principle for the Liquid Hydrogen Eddy-Current Rig is the same as for the Liquid Nitrogen Rig. The main difference being the liquid medium.

Test Apparatus

The rig (fig. 13) consists of a stainless steel rotor which is supported vertically by two angular contact bearings. The rotor has three disks which simulate the rotor masses of typical cryogenic turbopumps. The lower bearing is supported by a squirrel cage so that the bearing mount stiffness can be controlled. The copper conductor which is fastened to the flexible rotor bearing support vibrates when the rotor passes through a critical speed. The shaft is driven by a air turbine and operates at speeds from 0 to 40 000 rpm. The first two shaft critical speeds are at 11 000 and 22 000 rpm. The lower bearing and damper assembly are enclosed in a vessel which contains liquid hydrogen. Thus the bearing and damper assembly are submerged in liquid hydrogen during tests. The liquid hydrogen maintains the damper temperature at −424 °F and simulates temperatures found in cryogenic turbopumps.

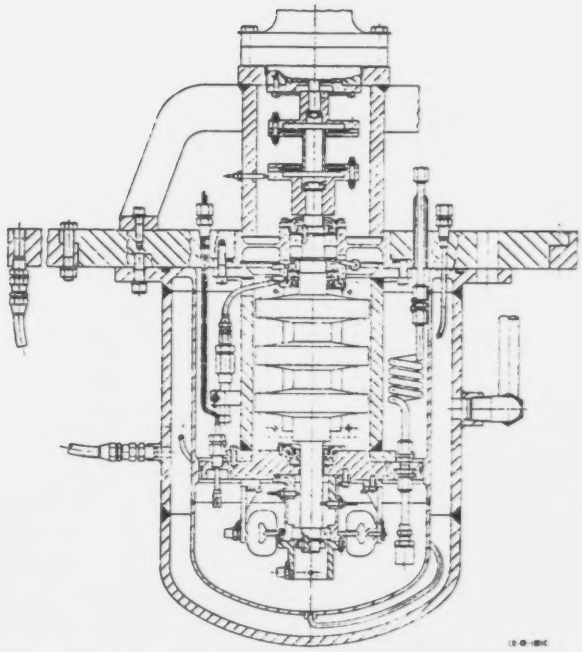

Figure 13.—Test apparatus for passive eddy-current damping in liquid hydrogen.

A controlled rotor vibration is induced by unbalancing the center rotor disk with set screws placed near the outside diameter of the disk. Two proximity probes located near the plane of the lower bearing measure the shaft orbit. Two accelerometers mounted at the lower end of the bearing housing measure bearing housing accelerations in orthogonal directions. Data from the proximity probes and accelerometers and a synchronous (1/revolution) pulse can be recorded on tape. These data can then be processed in a computer which gives synchronous rotor response and phase angle at the lower bearing.

Dynamic Spin Rig

Research leader: Erwin H. Meyn
Rig location: Building 5, Test Cell CW-18
Control room: Room CW-20

Purpose

The purpose of the Dynamic Spin Rig is to obtain steady-state and vibratory measurements of turbomachinery and propfan bladed disks rotating in a vacuum. The rig has been used to study shrouded compressor blades, swept twisted plates, propfan blades, and friction dampers.

Principle

The test article is mounted on a rotor with a vertical axis in a chamber that can be evacuated of air. An air turbine is used to spin the test article. The test article can be dynamically forced by shaking the shaft with two electromagnetic shakers. The two shakers apply oscillatory axial forces (220 N, 50 lbf) or transverse moments to the rotor shaft through a thrust bearing. Additional excitation can be provided by an air jet impinging on the blades. In the past, the blade responses were monitored by an array of optical probes (fig. 14) as the rotor speed was varied.

Test Apparatus

The vacuum chamber (fig. 14) is a fabricated steel cylindrical weldment with a 1-in.-thick shell, a 4-ft inside diameter, and a 3.8-ft height. It stands with its centerline vertical on three legs so the bottom of the vacuum chamber is 2 ft above the floor. A fragmentation shield, constructed of mild steel, fits down into the vacuum tank in the region of the rotor plane-of-rotation. A vacuum pump system can evacuate the tank volume of 48 ft^3 to a pressure of less than 0.75 mm of Hg. The vacuum reduces the torque required to drive the rotor, and it makes aerodynamic effects negligible. A Barbour Stockwell 411 air turbine, supplied by a regulated 125-psig service air supply, is used to drive the rotor. Rotors up to 24.5 in. in diameter can be spun up to 18 000 rpm. A remotely operated turbine air brake is used to stop the rotor.

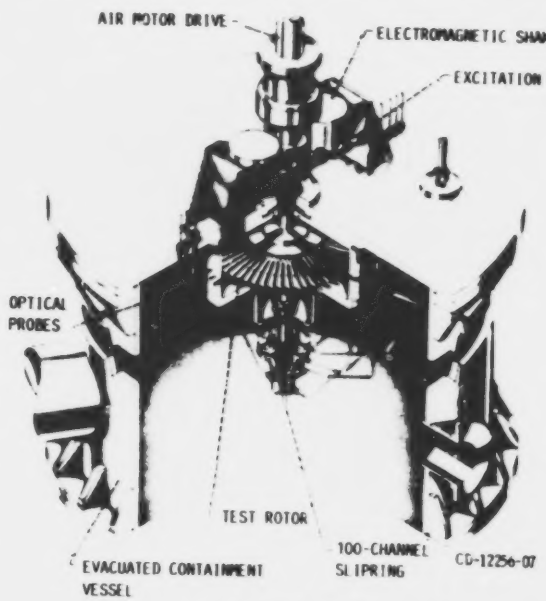

Figure 14.—Cutaway drawing of spin rig.

A dynamic balance of the shaft/rotor is done prior to installation in the facility. The test rotor has a main drive shaft supported on two single-row ball bearings. One bearing is at the top cover plate, and the other is at the lower end of the drive shaft. The lower bearing is installed in the bottom end of a cylindrical structure that surrounds the rotor. The rotor bearings are mounted on rubber supports which permit the rotor assembly to be driven in an axial oscillatory motion. Proximity probes are used to monitor transverse orbit signals of the rotating shaft. At the upper end of the drive shaft a torque coupling connects the shaft to a 4-in. Barbour Stockwell air turbine. The turbine is controlled by an electro-pneumatic servothrottle valve and is maintained to 0.5 percent of the specified full speed. The signal to the turbine speed controller is provided by a magnetic speed sensor and a 60 tooth gear located at the bottom of the drive shaft.

Below the drive shaft is a 100-channel Poly-Scientific slip ring assembly for the strain gage signals. The slip ring assembly is rated at 30 000 rpm and is cooled and lubricated by a mixture of high-boiling-point chlorofluorocarbon and a synthetic turbine oil. A seal system prevents leakage of the fluid into the vacuum chamber.

The spin rig is presently being used to measure SR3C-3 turboprop blade deflections. A laser blade deflection measurement system (fig. 15) consists of 0.5-mW randomly polarized lasers, power supplies, photodetectors, and amplifiers. The

Figure 15.—Turboprop laser setup.

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lasers are oriented such that the blades will interrupt the beams as the blades rotate. The photodetectors and the laser beams are on the opposite side of the blades. When the lasers impinge on the photodetectors a voltage is produced. As a blade interrupts the laser beams, the voltage from the photodetector drops to zero. This produces a square pulse signal. The width of the pulse produced by the blade interrupting the laser beam is proportional to the twist deflection. The system also uses a mirror mounted on the propeller hub or shaft to reflect a laser onto a photodetector to produce a 1/revolution (1P) signal. The position of the blade-produced square pulse with respect to the 1P signal is proportional to the bending deflection. The photodetector signals are recorded on magnetic tape and later digitized. The digitized data are then input to a computer program where deflections of the blades are computed.

A 2-ton trolley crane is used to transfer the test assembly from the buildup stand over to the vacuum tank. Two 6-ft-high tripod buildup stands (fig. 16) are used to assemble and disassemble the rotor system.

Additional information is given in references 2 and 6.

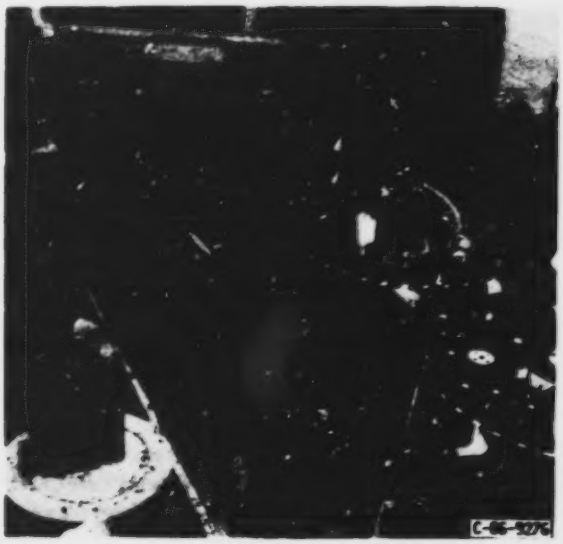

Figure 16.—Spin rig tank and buildup stands.

High Load/Thrust Bearing Damper Rig

Research leader: David P. Fleming
Rig location: Building 5, Test Cell CW-18
Control room: Room CW-20 A

Purpose

The purpose of this test rig is to test passive dampers that are designed to carry steady thrust loads and/or higher than normal rotating forces. The test condition could represent a condition similar to that when a turbine engine loses a blade.

Principle

A rotating load is applied to the test damper; the rotating load in turn is produced by unbalancing the shaft of the rig. Simultaneously, the test damper may have a steady thrust load applied from a pneumatic cylinder. Shaft and damper motions are measured by eddy current probes; damper load is measured by quartz load washers. These data, together with a phase reference signal, are used to calculate dynamic stiffness and damping of the test damper.

Test Apparatus

The test rig (fig. 17) sits on a steel table that is anchored to the test cell floor. The significant features of the rig are the 15-hp ac variable frequency drive motor, belt and pulley drive system, ball bearing slide, pneumatic cylinder, two oil supply systems, rotor assembly, and test damper housing. The drive motor has a maximum speed of 1750 rpm. The rig uses a poly V-belt system to drive the rotor to a maximum speed of 10 000 rpm. The dual oil supply system supplies lubrication to the bearings and to fluid dampers if required. The rotor

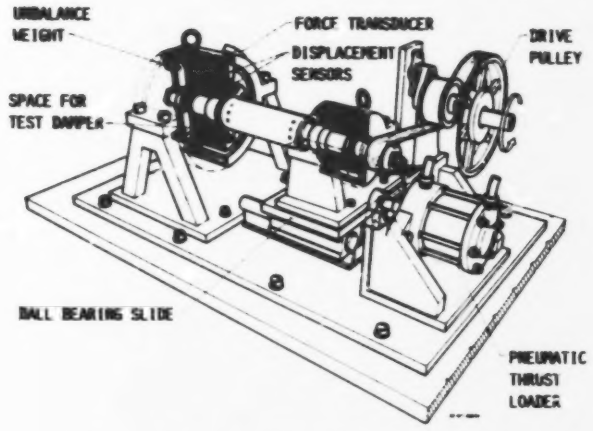

Figure 17.—High Load/Thrust Bearing Damper Rig.

amplitude data are collected by proximity probes, and the radial forces are monitored by four quartz force transducers. The quartz force transducers are mounted on the damper housing. The analog data are converted to digital data by a digital vector filter and transmitted to a digital computer where it is reduced and stored.

The thrust load and unbalance are determined and applied to the rotor and test damper. This condition is used to excite the test damper. A pneumatic cylinder, with a maximum load capacity of 1000 lb, applies a steady axial load at the drive end of the rotor. The axial load is transmitted through a thrust bearing assembly, which is supported on a ball bearing slide, and then to the rotor. There is no steady radial load applied to the rotor. The rotating load generated by the known unbalance is monitored at the damper. The collected test data are input to an online computer which converts the data into stiffness and damping values of the test damper.

Hypersonic Engine Seal Rig

Research leader: Bruce M. Steinetz
Rig location: Building 49, Furnace Room
Control room: Same

Purpose

The purpose of the Hypersonic Engine Seal Rig is to measure the performance of various engine seal concepts under engine simulated temperatures (up to 1800 °F) and pressures (up to 100 psi). Candidate seal leakage will be measured as a function of applied pressure differential, seal preload, and seal counterface surface parameters (waviness and roughness). Also, actuation (i.e., frictional drag) forces will be measured to assess the benefits of advanced high-temperature seal lubricants.

Principle

The test seal, fabricated in an 8-in.-diameter ring configuration, is mounted to the face of an axial loading piston (fig. 18). Testing engine seals in this unique ring configuration, instead of in a linear configuration, eliminates the need for any other high temperature supporting seal that would introduce error in leakage measurements. The seal mates against an interchangeable counterface surface and seals a high-temperature, pressurized chamber of air. This pressurized chamber is immersed in a cylindrical tungsten mesh furnace and heated up to 1800 °F. Seal leakage is accurately measured by a flowmeter in series with the incoming air (the centermost axial tube in the figure). Seal chamber pressure is measured by a static pressure tap protruding into the chamber (not shown in figure).

Seal normal loads are applied with the hydraulic actuator and measured with a load cell. Seal frictional forces are determined by measuring the torque required to rotate the seal about its axis. The seal friction coefficient is then calculated by dividing drag torque by the product of normal force and seal radius. New seal concepts ranging from a static high-temperature metal C-ring seal to an advanced dynamic flexible ceramic rope seal will be tested in this facility.

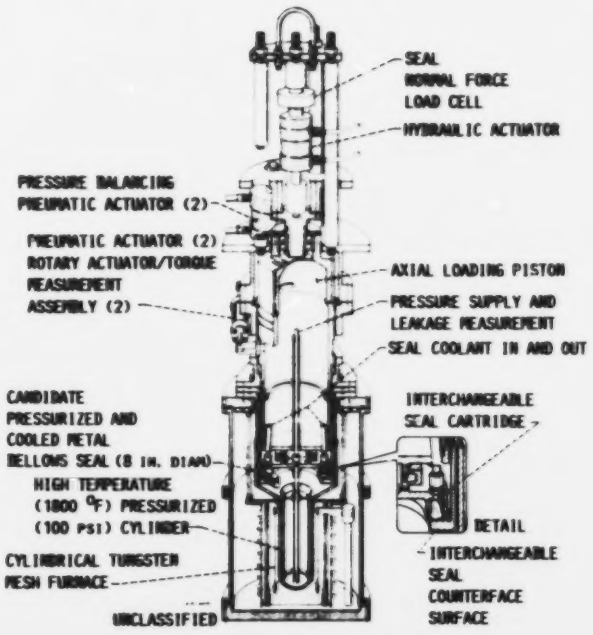

Figure 18.—Hypersonic Engine Seal Test Rig.

Test Apparatus

The main elements of the rig include the seal and axial loading piston; a high-temperature sealing counterface surface and a high pressure cylinder; a tungsten mesh furnace that operates in the stainless steel vacuum chamber; and the 60-kW furnace power supply. The rig, when completed in the spring of 1988, will stand approximately 12 ft high.

The rig uses shop air to supply the estimated 0.15 lb/sec airflow at the 100 psi required for testing. Shop air also supplies the pressure balancing pneumatic actuators to offset or balance the chamber pressure acting across the bottom face of the piston. Bottled air or nitrogen is used to supply seal coolant/pressurization when pressures in excess of shop air are required. Heated exhaust gases are cooled by air-water heat exchangers.

All supporting dynamic seals required for rig operation are an elastomer type and are deliberately located in cool areas of the facility and are further cooled by water jackets.

Additional information is given in reference 7.

Vacuum Roller Contact Rig

Research leader: Douglas A. Rohn
Rig location: Building 5, Test Cell CW-14
Control room: Same

Purpose

The purpose of the Vacuum Roller Contact Rig is to obtain performance and wear information on roller pairs acting in traction contact under thermal, vacuum, or other gas environment conditions. The rig will be used to study materials, coatings, dry lubricants, and contact geometry conditions.

Principle

Test roller specimens are supported by shafts in rotary motion feedthroughs within a vacuum chamber. A variable speed dc motor and magnetic particle brake provide rotation (500 rpm) and torque loading (600 in.-lbf). The output shaft includes a transducer to measure torque and axial thrust. The input shaft is gimballed to provide roller normal loading, traction force measurement, and roller skewing. The chamber can be evacuated or filled with another gas. Electric heaters and cryo-coolers provide thermal conditions.

Test Apparatus

A schematic drawing of the rig is shown in figure 19. The cylindrical stainless steel vacuum chamber has a 12-in. outside diameter and is 27.5 in. high. Attached to the bottom is a diffusion pumping system to provide operating pressures in the 1 μtorr range. The chamber is also capable of being backfilled with other gaseous environments. The entire rig is mounted on an optical-grade table and an air-piston vibration isolation system.

Test roller specimens are mounted on shafts which are supported by ferrofluidic feedthroughs. Both external and internal contact geometries can be tested. A variable speed permenant magnet motor and gearbox provide power and rotation, up to 500 rpm and 1.5 hp. The input feedthrough, shaft, and motor are gimballed about a vertical and horizontal axis so that roller traction normal loads (up to 200 lb) and tractive forces can be applied and/or measured by load cells.

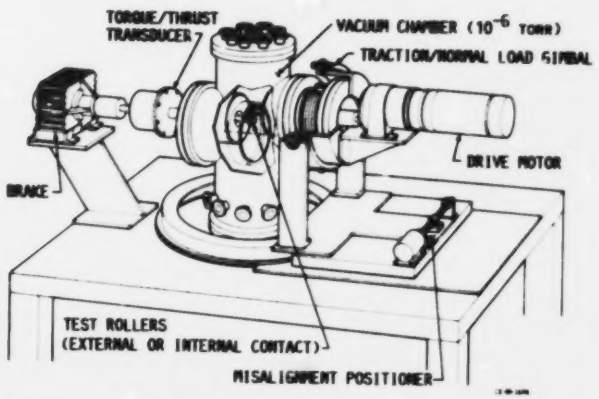

Figure 19.—Space-simulated roller contact performance tester.

The entire input assembly, including gimball, can be rotated about the vertical centerline of the tank, which is also the specimen contact centerline. This allows the rig to be run with skewed specimen axis to measure traction performance via the side-slip method.

The output shaft assembly consists of two concentric shafts. The outer shaft provides scaling and support to the inner shaft, which is mounted in combination linear/rotating bearings. Output specimen torque and axial thrust are transmitted via this shaft to a torque/thrust cell to measure traction torque transfer and skewed-axis thrust forces. Power is absorbed in a magnetic particle brake, up to 600 in.-lb torque over the full speed range. A belt-driven flywheel is also available to absorb transient torque changes for which the brake may not react quickly enough. Optical encoders provide speed signals on both input and output. To provide thermal conditions on the specimens, a heating and cooling system is being designed for roller temperatures from −40 to +100 °C. A data acquisition system obtains and records pertinent data for both long-term wear tests and the shorter duration generation of traction force verses creep curves.

The vacuum roller contact rig is presently being assembled to measure the traction characteristics in air and vacuum of steel rollers ion plated with gold for space mechanism and robotics use. Preliminary wear data will also be obtained. Future testing will include hard coatings and other ion implantations on metallic rollers, as well as polymer/elastomer coatings on solid polymer rollers.

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Bench Test Laboratory

Research leader: Erwin H. Meyn
Location: Building 5, Room CW-20-B

Purpose

The Bench Test Laboratory provides an environment for conducting vibration experiments on small turbomachinery components such as a single blade. These experiments are nonrotating. The purpose is to determine stiffness, natural frequencies, mode shapes, and damping characteristics of these components.

Principle

Figure 20 shows a fixture that is used to mount the test article to an isolation table for modal analysis. Figure 21 is a schematic of a modal analyzer experiment. Impact, random, and sine sweep excitations are used on the test article. The forcing is normally provided magnetically. Normally the output transducer is a miniature accelerometer. Standard modal methods are then used to measure the natural frequencies, mode shapes, and damping of the component. Standard modal analyzers are used to operate on the raw output data. In some cases, e.g., our friction damper experiments, unique experimental techniques are developed.

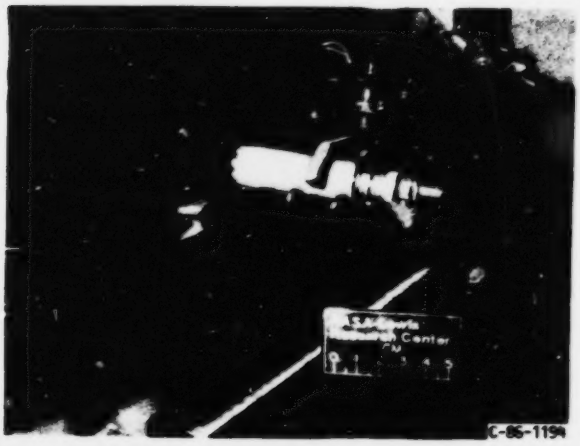

Figure 20.—Modal analyzer experimental apparatus.

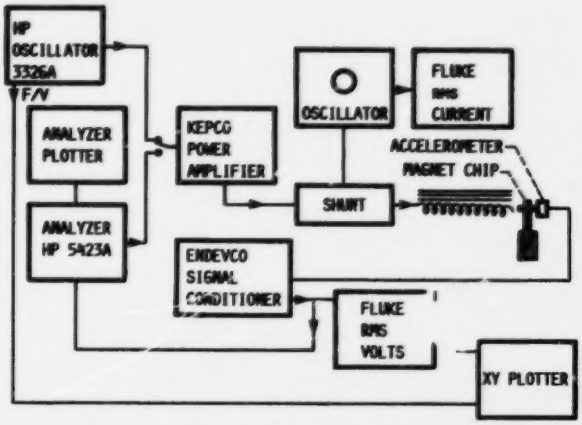

Figure 21.—Modal analyzer experimental schematic.

Test Apparatus

The most commonly used pieces of equipment are the following:

Isolation table (4 ft × 6 ft)
Electromagnetic driving source
Hewlett Packard spectrum analyzer (3580A)
Hewlett Packard 7046A x-y recorder
Fluke 1953A counter timer
Kepco bipolar operation power supply/amplifier
Fluke 8920A true rms voltmeter
Saber VII magnetic tape recorder
Endevco vibration amplifier (model 6630)
Accelerometers HU38 M2222 C
Tektronix 7603 oscilloscope

Magnetic Bearing and Suspension Laboratory

Research leader: Gerald V. Brown
Rig location: Building 5, Room CW-20-C
Control room: Same

Purpose

The purpose of the Magnetic Bearing and Suspension Laboratory is to investigate and improve methods of magnetically suspending spinning shafts and other systems that require position control and vibration suppression.

Principle

Magnetic suspension has been used for a number of years because it offers suspension without physical contact and the ability to induce vibration damping in a controlled way. Some types of magnetic suspension devices are inherently stable and some inherently unstable without electronic controls. Advances in electronic controls now make possible magnetic suspension systems with very accurate position control and a high degree of vibration suppression.

Test Apparatus

The laboratory is in its initial buildup phase. The intended work includes research on each of the three principle parts of magnetic suspension systems: the magnetic actuators, various types of sensors (displacement, velocity, and acceleration), and the control power electronics.

The basic facilities presently available include the large vibration isolation table in the bench test lab on which to mount experimental setups, a coil winding machine for winding actuator coils, an experiment buildup area, an initial stock of basic electronics equipment (such as power supplies and power amplifiers, oscillators, signal amplifiers, oscilloscopes, proximity probe, and accelerometer electronics), and adjacent machine and electronics shops.

Progress to date includes the fabrication of magnetic actuators of both the Lorentz and magnetic circuit type (with force capabilities of up to 100 lb). Position and velocity sensors have also been fabricated and used in the inital demonstrations of magnetic control in a single degree of freedom system. The system was first suspended by a mechanical spring and then suspended magnetically. In both cases controlled positioning with magnetic stiffness about the control point and magnetic damping of disturbances was shown.

Planned work includes the development of higher performance actuators, development of position sensors for systems with large travel, the suspension of multiple degree of freedom platforms to provide low acceleration mounts for sensitive experiments, the suspension of a spinning shaft by magnetic actuators, and the development of analog and digital control circuitry with improved performance and reduced size and weight.

Robotics Laboratory

Research leader: Douglas R. Rohn
Laboratory location: Building 5, Test Cell SW-17
Control room: SW-15

Purpose

The purpose of the Robotics Laboratory is to evaluate advanced mechanical and structural concepts for sp ce robotic applications. The ability to control robot base reactions, manipulate at microgravity acceleration levels, and optimize end-effector trajectories will be verified. The laboratory will be used to study experimentally roller-driven manipulation joints, flexibility, momentum compensation, and system dynamics.

Principle

The basic concept is shown in figure 22. Reaction-limited manipulation will be studied via a 4 degree-of-freedom (DOF) manipulator. The manipulator will be mounted on a sensing platform which is capable of measuring all six reaction components over a 10 000 to 1 range. Future robot system rig(s) will be of increasing complexity.

Test Apparatus

The six-component force/moment transducer is a stiff platform supported by strain-gaged elastic elements. Their geometry is such that, by suitable algebraic manipulation and calibration, all six robot arm reaction components can be measured. The platform is designed for static and dynamic loads of 100 lbf each and a 1300 in.-lb moment. A high-speed data acquisition system will record dynamic loading during manipulator arm motion.

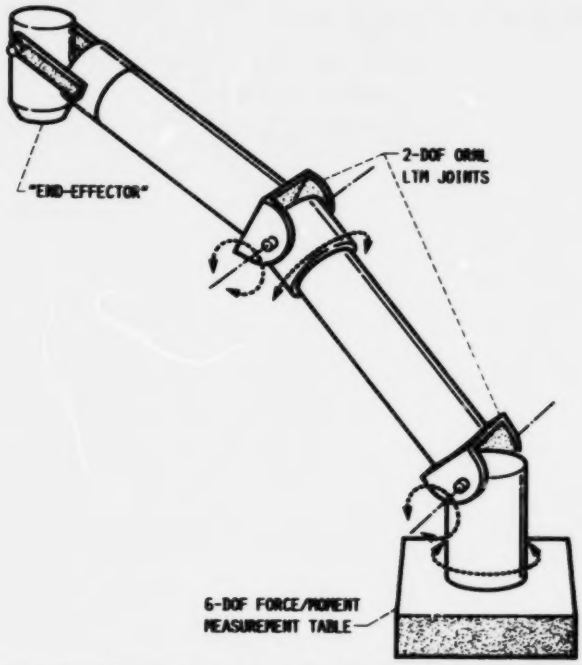

Figure 22.—Microgravity/reaction compensation manipulator experiment concept.

The test robot arm consists of two joints from the ORNL/LaRC Laboratory Telemanipulator (LTM) system. Each joint is a 2-DOF mechanism driven by two dc motors via a roller traction drive differential. Besides being used as a vehicle for reaction compensation measurements, this simple robot arm will allow measurements of smooth, roller-driven joints and their effects on microgravity-level manipulation.

Transputer Laboratory

Research leader: David C. Janetzke
Location: Building 23, Room W118

Purpose

The Transputer Laboratory provides a versatile parallel processing computer for testing new or modified algorithms in structural dynamic analysis.

Principle

Present-day supercomputers approach the practical limit for speed in the computation of serial algorithms. To achieve faster computation speed, a problem must be dissected into independent processes and solved simultaneously on a system of multiple processors. Parallel processing systems are now becoming available with a variety of network architectures for data transfer between processors. Many of these architectures can be simulated with a network of unique microcomputers called transputers. To evaluate the effectiveness of these different systems for structural dynamic analysis, algorithms are implemented on a transputer-based system.

Apparatus

A transputer is a single VLSI chip containing a microprocessor, on-chip memory, and four serial input/output data links for connection to other transputers. It is a product of the INMOS Corporation. The transputer design is optimized to run Occam, a language developed for programming concurrent processes. A block diagram of a 32-bit transputer with floating point coprocessor is shown in figure 23.

The parallel processing system is a desktop-size system of transputers hosted by an NCR PC8 personal computer. The PC contains a board with one 32-bit transputer and 2 Mbyte DRAM (dynamic random access memory). It serves as the host for a system of interconnected transputers and provides keyboard input, screen output, and disk I/O.

A total of 68 INMOS transputers are contained in two small cabinets (fig. 24). One cabinet has 10 boards with four 32-bit transputers on each board. Each board also has 256 Kbyte DRAM per transputer. The other cabinet has 3 boards with 9 16-bit transputers on each board and a graphics board. The 16-bit transputer board has 54 Kbytes DRAM for one transputer and 8 Kbytes DRAM for each of the others. The graphics board has a 32-bit transputer, 512 Kbytes of dual ported screen memory and 512 Kbytes of program memory. All of these transputers can be interconnected in a large variety of ways. The graphics board delivers a medium resolution RGB signal to the color monitor (fig. 24).

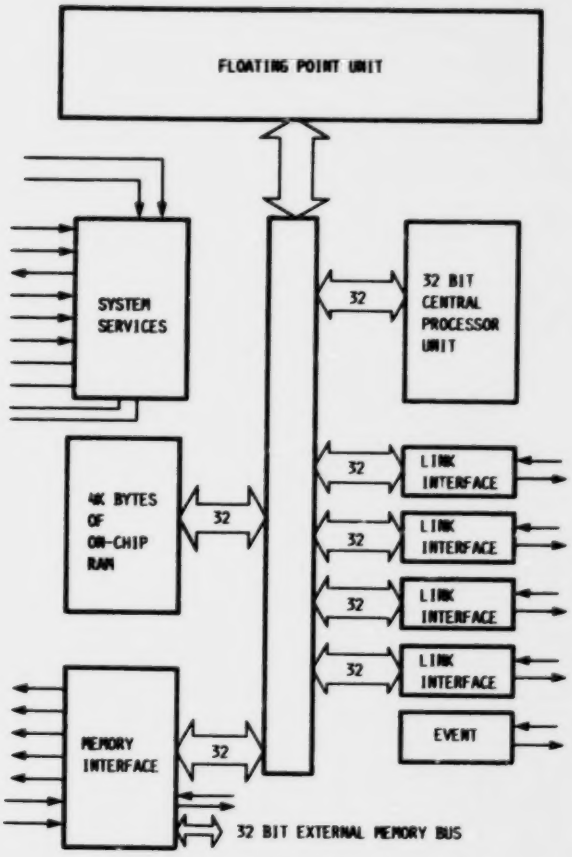

Figure 23.—Block diagram for T800 transputer.

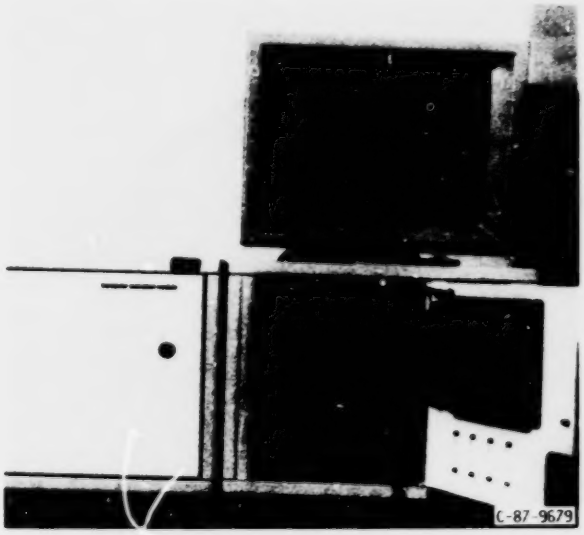

Figure 24.—Transputer-based parallel processing system.

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Software development is done on the PC. A high-level language called Occam is primarily used to implement the unique concurrent processing and communication features of the INMOS transputers. Compilers are also available for other high-level languages such as C, FORTRAN 77, and Pascal.

Several types of transputers are available. The T212 transputer has a 16-bit microprocessor with 2 Kbytes of on-chip memory. The T414 transputer has a 32-bit microprocessor with 2 Kbytes of on-chip memory and is capable of about 100 000 32-bit floating point operations per second. The T800 transputer has a 32-bit microprocessor with 4 Kbytes of on-chip memory and a floating point coprocessor. The T800 transputer is capable of sustaining 1.5 million floating point operations per second.

Concluding Remarks

This report briefly describes 10 test rigs and 4 laboratories of the NASA Lewis Research Center's Structural Dynamics Branch. The rigs are associated with research in the areas of aeroelasticity, vibration control, dynamic systems, and computational structural methods.

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15. Supplementary Notes

Gerald T. Montague, Sverdrup Technology, Inc., (Lewis Research Center Group), NASA Lewis Research Center, Cleveland, Ohio 44135; Robert E. Kielb, NASA Lewis Research Center.

16. Abstract

The NASA Lewis Research Center Structural Dynamics Branch conducts experimental and analytical research related to the structural dynamics of aerospace propulsion and power systems. The contents of this paper deal with the experimental testing facilities of the branch. Presently there are 10 research rigs and 4 laboratories within the branch. These facilities are described along with current and past research work.

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